NASA Technical Paper 1657

Operating Characteristics of a High-Speed, Jet-Lubricated 35-Millimeter-Bore Ball Bearing With a Single-Outer-Land-Guided Cage

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APRIL 1980

NASA

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NASA
National Aeronautics
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Scientific and Technical
Information Office

1980

Summary

Parametric tests of a 35-mm-bore, angular-contact ball bearing with a single-outer-land guided cage were conducted on a high-speed, high-temperature bearing tester. The bearing had an unmounted nominal contact angle of 24°. Provisions were made for jet lubrication of the bearing and for outer-ring cooling. Test conditions included thrust loads of 667 and 1334 N (150 and 300 lb) and a combined radial and thrust load of 222 and 667 N (50 and 150 lb), respectively. Nominal shaft speeds ranged from 28 000 to 72 000 rpm, with an oil-in temperature of 394 K (250° F). Lubricant oil was jet fed to the test bearing at flow rates ranging from 76 to 1894 cm^3/min (0.02 to 0.50 gal/min) and at a jet velocity of 20 m/sec (66 ft/sec). The lubricant, neopentylpolyol (tetra) ester, met the MIL–L–23699 specifications.

A 35-mm-bore, angular-contact ball bearing with jet lubrication was successfully operated to 2.5 million DN at a maximum thrust load of 1334 N (300 lb) and a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial.

Bearing temperatures increased with shaft speed and decreased with increasing lubricant flow to the bearing. The inner ring was generally cooler than the outer ring over the speed and flow ranges tested. A 1334-N (300-lb) thrust load produced higher temperatures than either a 667-N (150-lb) thrust load or a combined 667-N (150-lb) thrust and 222-N (50-lb) radial load, although the effect of load on temperature was relatively small. When outer-ring cooling was used to thermally balance the inner and outer rings of the bearing, higher operating speeds could be achieved with lower lubricant flow rates, and this resulted in lower operating temperatures.

Bearing power loss increased as the lubricant flow rate, speed, or load was increased. The highest thrust load, 1334 N (300 lb), caused the greatest amount of heat rejection to the lubricating oil, followed by the 667-N (150-lb) thrust load and then the combined 667-N (150-lb) thrust, 222-N (50-lb) radial load.

The increase in percent cage slip with increasing lubricant flow rate was minimal for all speed and load conditions tested. Percent cage slip decreased significantly when the thrust load was doubled, but it increased appreciably with speed for the lubricant flow rates and loads tested.

The oil-in temperature did not appreciably change the basic characteristics of the speed-versus-outer-ring-temperature curves when compared with data from another study employing bearings of similar bore size but with room-temperature inlet oil.

Introduction

Mainshaft bearings in present-production, large gas-turbine engines perform satisfactorily at a DN range to 2.3 million. (DN is defined as the speed of the shaft in revolutions per minute multiplied by the bearing bore in millimeters.) Small-bore bearing experiments in the past have generally concentrated on tests at DN values at or below 1.8 million. Reference 1 reports on 20-mm-bore ball bearings at DN values to 1.8 million and focuses on the proper techniques for scavenging test-bearing lubricating oil in order to reduce bearing operating temperature. Reference 2 deals with the friction and surface damage aspects of high-speed ball bearing operation to 1.0 million DN. Reference 3 reports on the minimum oil requirements of 30- and 75-millimeter-bore ball bearings at DN values to 0.975 million. Reference 4 thoroughly investigates small-bore (30 mm) ball bearing operation, where such factors as cage friction, oil jet velocity, oil jet positioning, and cage design are of principal interest. In all these tests the oil for bearing lubrication was not preheated, and the operating speeds were below 2.5 million DN.

Large-bore ball and roller bearings have been successfully tested to 3.0 million DN (refs. 5 to 8). However, in these tests the lubricant was fed to the bearings through radial holes in the inner ring. Because of the dimensional limitations of the inner ring in smaller bore bearings, the fabrication of radial holes and axial grooves for lubricant passages through the inner ring can become complex and cost restrictive. In these circumstances jet lubrication is the more practical method of bearing lubrication.

Advanced engines in the small class (1- to 10-lb/sec total airflow) require bearings that operate in the 2.5-million-DN range at high temperatures in order to achieve the performance objectives set by the U.S. Army or programs such as STAGG (Small Turbine Advanced Gas Generator) and UTTAS (Utility Tactical Transport Aircraft System). The bearing designs and lubrication techniques used for these engines must be refined and optimized for reliable performance and long life.

In pursuing this task the major factors affecting the lubricant flow to and through a 35-mm-bore, angular-contact bearing with a single-outer-land-guided cage were investigated. The objectives of this study were to determine the effects of the following parameters: (1) shaft speed, (2) thrust and combined thrust and radial loading, (3) lubricant flow rates, and (4) outer-ring cooling.

The bearing had a nominal (unmounted) contact angle of 24°. Provisions were made for jet lubrication of the bearing and for outer-ring cooling. Test conditions included thrust loads of 667 and 1334 N (150 and 300 lb) and a combined radial and thrust load of 222 and 667 N (50 and 150 lb), respectively. Nominal shaft speeds were 28 000 to 72 000 rpm, with an oil-in temperature of 394 K (250° F). Lubricant was jet fed to the bearing at flow rates from 76 to 1894 cm^3/min (0.02 to 0.50 gal/min). Outer-ring cooling oil flow rates were 0 to 682 cm^3/min (0 to 0.18 gal/min) at 394 K (250° F) oil-in temperature. The lubricant, neopentylpolyol (tetra) ester, met the MIL-L-23699 specifications.

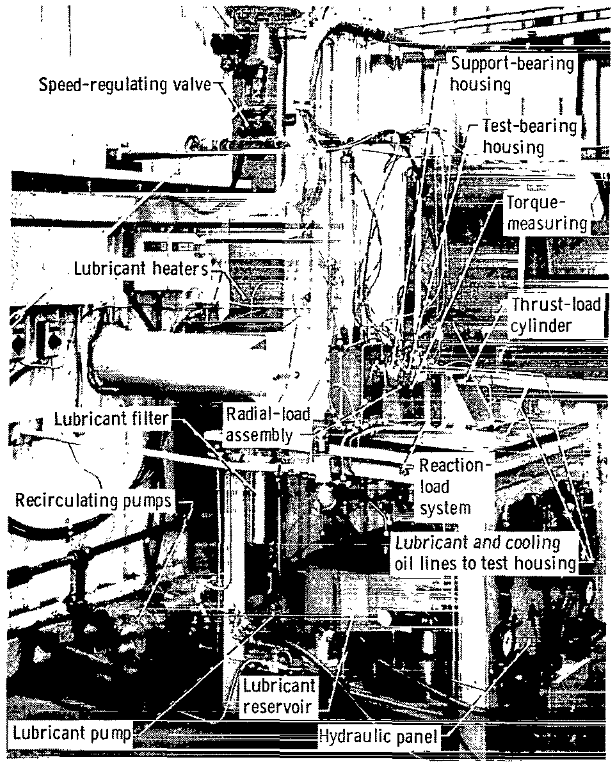


Figure 1. - High-speed, small-bore-bearing test machine.

Apparatus and Procedure

High-Speed Bearing Tester

A general view of the air-turbine-driven test machine is shown in figure 1. A sectional drawing is shown in figure 2. The shaft is mounted horizontally and is supported by two preloaded angular-contact ball bearings. The test bearing is assembled into a separate housing that incorporates the hardware for lubrication, oil removal, thrust and radial load application, and ball-pass frequency (cage speed) measurement. Test bearing torque is measured with strain gages located near the end of an arm that prevents the housing from rotating.

Thrust force is applied through a combination of a thrust needle bearing and a small roller support bearing in order to minimize test housing restraint during torque measurements. Radial load is applied to the test bearing through knife-edge bearings, which effectively minimize friction.

The test bearing was lubricated by two jets on the nonloaded side of the inner ring. The jet outlets, located approximately 3 mm (0.12 in.) from the face of the bearing, were aimed at the inner raceway. In separate tests, not reported herein, it was determined that a 20-m/sec (66-ft/sec) jet velocity insured the most efficient lubrication of the test bearing, and this velocity was used in all the tests reported. (Reference 4 reports a similar efficiency with a 20-m/sec (66-ft/sec) jet velocity over other velocities.) Cooling oil was supplied to the outer ring by means of holes and grooves in the bearing housing, as shown in figure 2.

Shaft speed (inner-ring speed) was measured with a magnetic probe. Ball-pass frequency (cage speed) was measured with a semiconductor strain gage mounted in a cavity of the housing and was displayed on a spectrum analyzer.

Two thermocouples were assembled in the shaft so that the centrifugal force would push them against the test-bearing inner ring. Temperature readings were transmitted with a rotating telemetry system mounted on an auxiliary shaft at the air-turbine end of the test machine. Outer-ring temperatures were obtained by two thermocouples installed in the test-bearing housing. One was located 45° from the center of the radial load zone of the bearing; the other was positioned 180° from the first. For accurate measurement of oil-in and oil-out temperatures, thermocouples were placed directly in the housing fittings of the lubricating jets and in the oil discharge reservoir, respectively, as shown in figure 2.

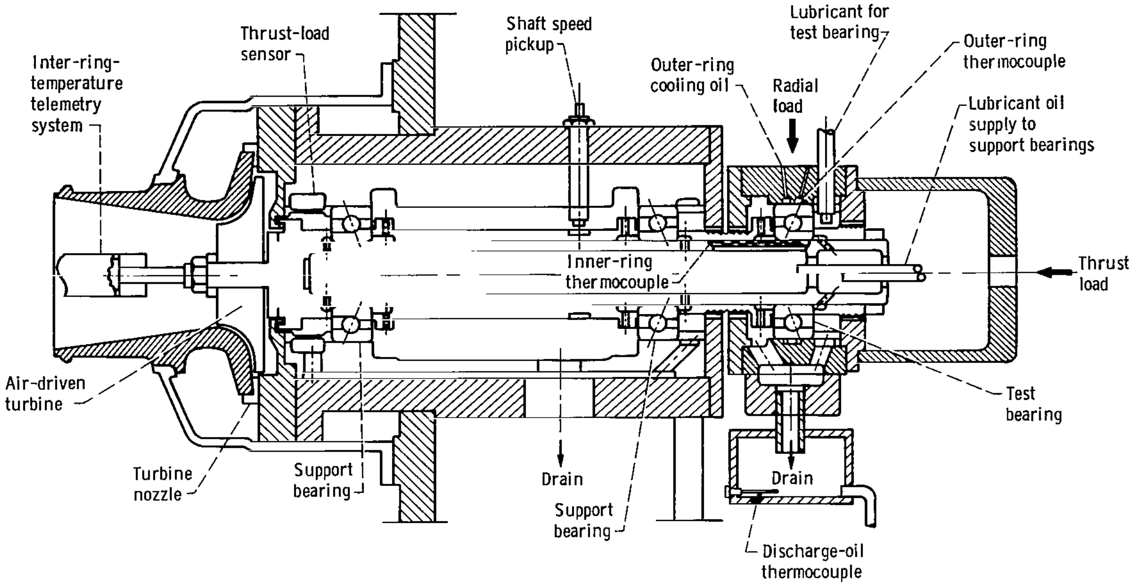


Figure 2. - Schematic of high-speed, small-bore-bearing test machine.

Test Bearing

The test bearing (fig. 3) was an ABEC-grade, 35-mm-bore, angular-contact ball bearing with the cage riding on one outer-ring land (fig. 4). The bearing contained 16 balls with a nominal 7.14-mm (0.281-in.) diameter. The inner and outer rings, as well as the balls, were manufactured from consumable-electrode, vacuum-melted AISI M-50 steel. The nominal hardness of the balls and rings was Rockwell C62 at room temperature. The cage was made from AISI 4340 steel (AMS 6415) heat treated to Rockwell C28 to C36 hardness. It was completely coated with a 0.0203- to 0.0381-mm (0.0008- to 0.0015-in.) thickness of silver plate (AMS 2412). The cage balance was within 0.05 g-cm (7×10^{-4} oz-in.). Additional specifications are shown in table I.

Lubricant

The lubricant used for the parametric studies was a neopentylpolyol (tetra) ester. This type II oil is qualified to the MIL-L-23699 specifications as well as to the internal oil specifications of most major aircraft engine producers. The major properties of the lubricant are presented in table II.

Test Procedure

After the test machine had been warmed by recirculating heated oil and the torque-measuring system had been calibrated, a 667-N (150-lb) thrust load and a 1894-cm^3/min (0.50-gal/min) lubricant flow rate were applied. The shaft speed was then slowly brought up to a nominal 28 000 rpm. When the bearing and test machine temperatures stabilized (after 20 to 25 min), the oil-in temperature and lubricant flow rates were set and the speed was increased to the desired value.

A test series was run by starting at the lowest nominal speed, 28 000 rpm, and progressing through 47 000, 65 000, and 72 000 rpm before changing the lubricant flow rate. At each speed and flow condition a separate test was run during which the outer-ring cooling oil flow was adjusted to achieve equal inner- and outer-ring temperatures. Five lubricant flow rates of 76 to 1894 cm^3/min (0.02 to 0.50 gal/min) were used, with a nominal jet velocity of 20 m/sec (66 ft/sec).

After these test runs were completed, other tests were performed to determine the effects of

(1) Increasing the thrust load to 1334 N (300 lb)
(2) Adding a nominal 222-N (50-lb) radial load to the 667-N (150-lb) thrust load

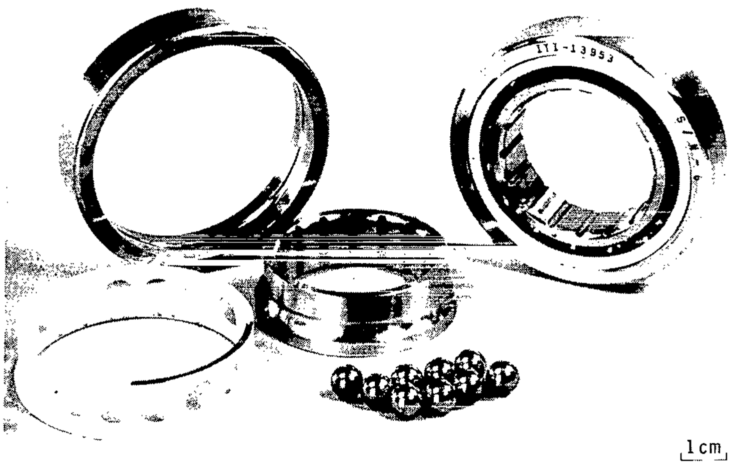


Figure 3. - 35-Millimeter-bore, angular-contact, high-speed ball bearing.

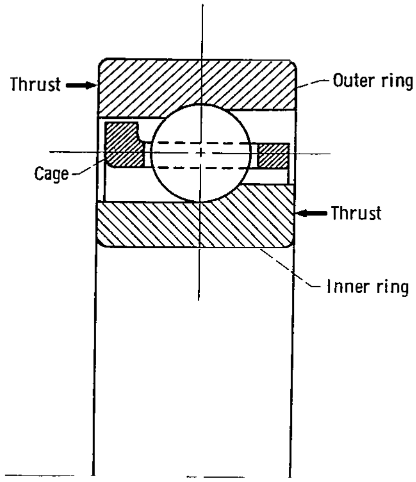


Figure 4. - Angular-contact ball bearing with a single-outer-land-guided cage.

TABLE I. - TEST BEARING SPECIFICATIONS

Bearing dimensions, mm (in.):	
Bore	35 (1.3780)
Outside diameter	62 (2.4409)
Width	14 (0.5512)
Cage specifications:	
Diametral land clearance, mm (in.)	0.406 (0.016)
Diametral ball-pocket clearance, mm (in.)	0.660 (0.026)
Material	AISI 4340 (AMS 6415) silver plated
Hardness	RC 28-36
Race conformity, percent:	
Inner	54
Outer	52
Assembly:	
Internal radial clearance, mm (in.)	0.074 (0.0029)
Contact angle, deg	24

TABLE II. - PROPERTIES OF TETRAESTER LUBRICANTS

Additives	Antiwear, corrosion and oxidation inhibitors, and antifoam
Kinematic viscosity, cS, at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flashpoint, K (°F)	533 (500)
Autogenous ignition temperature, K (°F)	694 (800)
Pourpoint, K (°F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt%	3.2
Specific heat at 372 K (210° F), J/kg K; Btu/lb °F	2140 (0.493)
Thermal conductivity at 477 K (400° F), J/m sec K; Btu/hr ft °F	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

For each of these conditions tests were run at nominal speeds of 47 000, 65 000, and 72 000 rpm at various lubricant flow rates from 76 to 1894 cm^3/min (0.02 to 0.50 gal/min).

If it became apparent during the course of testing that the conditions would result in predictable distress of the test bearing or test rig or generate a bearing temperature above 491 K (425° F), the test point was aborted or omitted.

Results and Discussion

Parametric tests were conducted in a high-speed bearing tester on a 35-millimeter-bore ball bearing with a single-outer-land-guided cage. Test parameters included load, shaft speed, lubricant flow rate, and outer-ring cooling oil flow rate. Tests were performed at thrust loads of 667 and 1334 N (150 and 300 lb) and at a combined 667-N (150-lb) thrust and 222-N (50-lb) radial load. Shaft speeds were a nominal 28 000 to 72 000 rpm. Test conditions included an oil-in temperature of 394 K (250° F). Oil was supplied by jet for bearing lubrication at flow rates of 76 to 1894 cm^3/min (0.02 to 0.50 gal/min). Outer-ring cooling flow rates of 0 to 758 cm^3/min (0 to 0.20 gal/min) at a 394 K (250° F) oil-in temperature were used in some tests. Over this entire range of test conditions, the bearing operated successfully without any visible damage to the components.

Effect of Lubricant Flow Rate and Load on Bearing Temperature

The effects of lubricant flow rate on bearing temperature at a 667-N (150-lb) thrust load, a 1334-N (300-lb) thrust load, and a combined 667-N (150-lb)

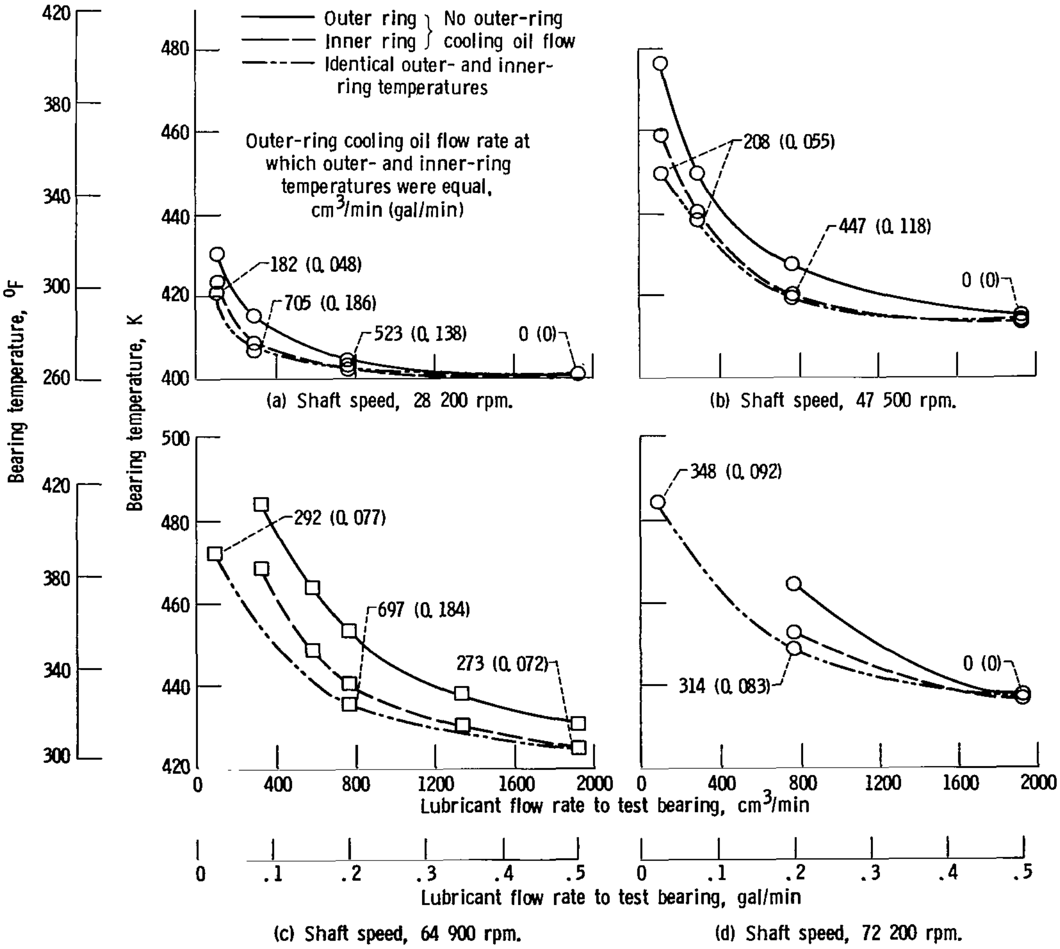


Figure 5. - Effect of lubricant flow rate on test-bearing temperature with and without outer-ring cooling for thrust load of 667 N (150 lb). Oil-in temperature, 394 K (250° F).

thrust and 222-N (50-lb) radial load are shown in figures 5, 6, and 7, respectively.

For those applications that may require (for best performance and optimal operating clearance) a minimal temperature gradient between the bearing inner and outer rings, tests were conducted to find the outer-ring cooling oil flow rate that produced a thermally balanced bearing, with equal inner- and outer-ring temperatures. These results are also shown in figures 5 to 7, with the required outer-ring cooling oil flow rate labeled at each data point.

Test-bearing outer- and inner-ring temperatures increased with shaft speed and decreased with increasing lubricant flow rate for all three load conditions with or without outer-ring cooling. Increasing lubricant flow rate beyond 758 cm^3/min (0.20 gal/min) resulted in less of a decrease in bearing temperature than was observed for flows under 758 cm^3/min (0.20 gal/min). The inner-ring temperature was generally lower than the outer-ring temperature for all speeds tested without outer-ring cooling. Generally, as the lubricant flow rate was increased, the inner- and outer-ring temperatures tended to approach equal values independent of speed or load conditions. Adding outer-ring cooling to thermally balance the inner and outer rings affected the outer ring much

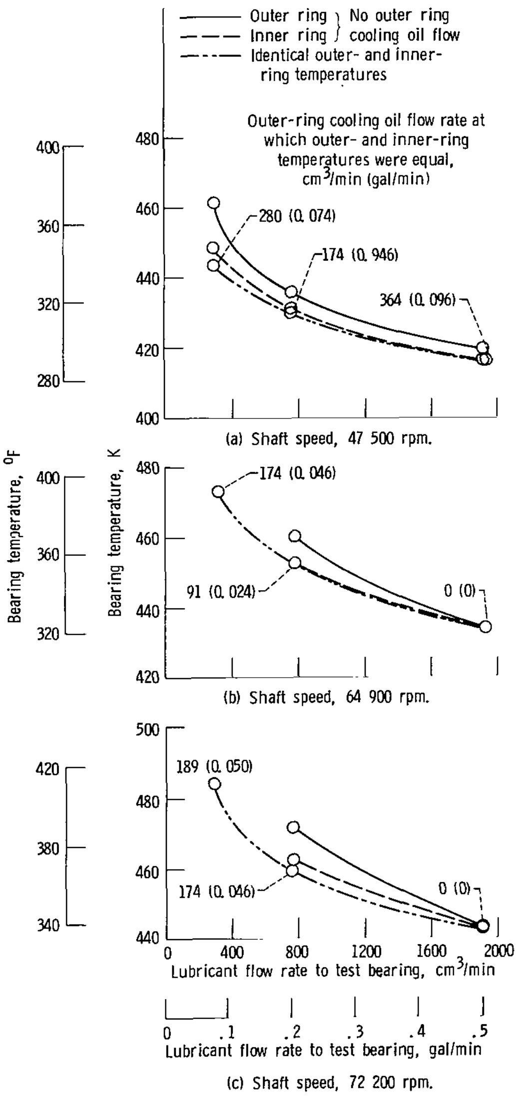


Figure 6. - Effect of lubricant flow rate on test-bearing temperature with and without outer-ring cooling for thrust load of 1334 N (300 lb). Oil-in temperature, 394 K (250° F).

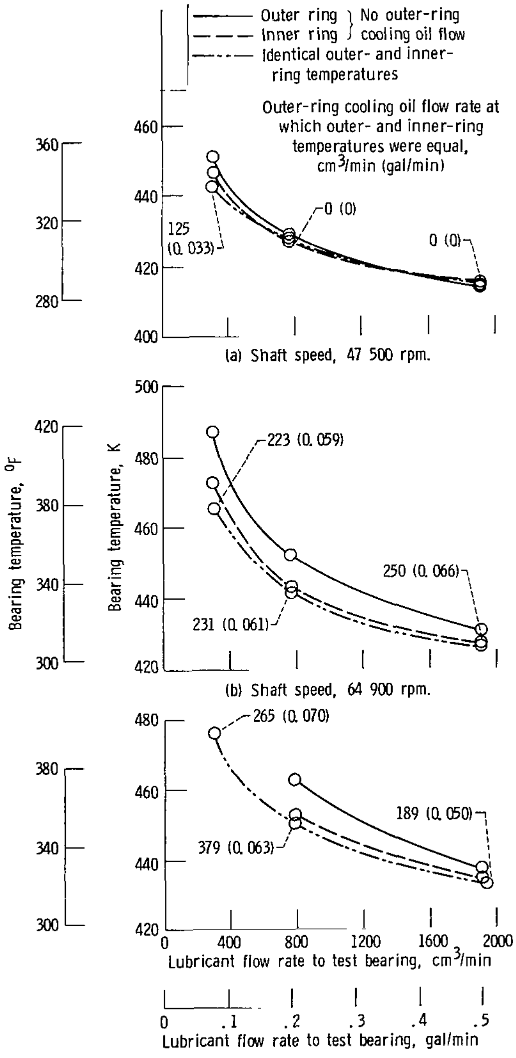


Figure 7. - Effect of lubricant flow rate on test-bearing temperature with and without outer-ring cooling for combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial. Oil-in temperature, 394 K (250° F).

more than the inner ring, as expected. Generally, outer-ring cooling resulted in a decrease in bearing operating temperature. The cooling oil flow rates required to produce a thermally balanced bearing ranged from 0 to 682 cm³/min (0 to 0.18 gal/min) and were generally less at the highest lubricant flow rate over the entire range of test conditions (figs. 5 to 7).

Bearing Temperature Versus Lubricant Flow Rate at Three Different Load Conditions

The effects of lubricant flow rate on bearing temperature are shown in figure 8 for three different load conditions: a 667-N (150-lb) thrust load, a 1334-N (300-lb) thrust load, and a combined 667- and 222-N (150- and 50-lb) thrust and radial load, respectively, at flow rates of 76 to 1894 cm³/min (0.02 to 0.50 gal/min). The outer-ring temperature data are shown in figure 8(a), and the inner-ring data in figure 8(b). Outer-ring temperatures for the 667-N (150-lb) thrust load and the combined-load tests show no appreciable difference throughout the speed range covered. However, outer-ring temperatures rose an average of about 7 kelvins (14 deg F) above the 667-N (150-lb)-thrust and combined-load data when a pure thrust load of 1334 N (300 lb) was applied.

The inner-ring temperatures (fig. 8(b)) reacted somewhat differently from the outer-ring temperatures. The higher thrust load, 1334 N (300 lb), produced the highest temperatures at the inner ring, as was the case with the outer ring. However, the combined load resulted in a greater increase in inner-ring temperature than in outer-ring temperature (fig. 8(a)) at the two lower shaft speeds of 47 500 and 64 900 rpm when compared with the 667-N (150-lb) thrust curves. At the maximum speed of 72 200 rpm, the inner- and outer-ring temperatures resulting from the combined-load tests were close to those for the 667-N (150-lb) thrust load.

Effect of Oil-In Temperature on Bearings of Similar Configuration

Data from figure 5 are compared in figure 9 with data for a similar ball bearing taken from reference 4. The main difference between these investigations is that the referenced data were obtained with a 30-mm-bore bearing and a 289 K (60° F) oil-in temperature, whereas the bearing data of figure 5 were obtained for a 35-mm-bore bearing and a 394 K (250° F) oil-in temperature. Although the curves for the 35-mm-bore bearing are naturally at a higher temperature level than those for the 30-mm-bore bearing, each pair of curves at equal lubricant flow rates parallel each other very closely.

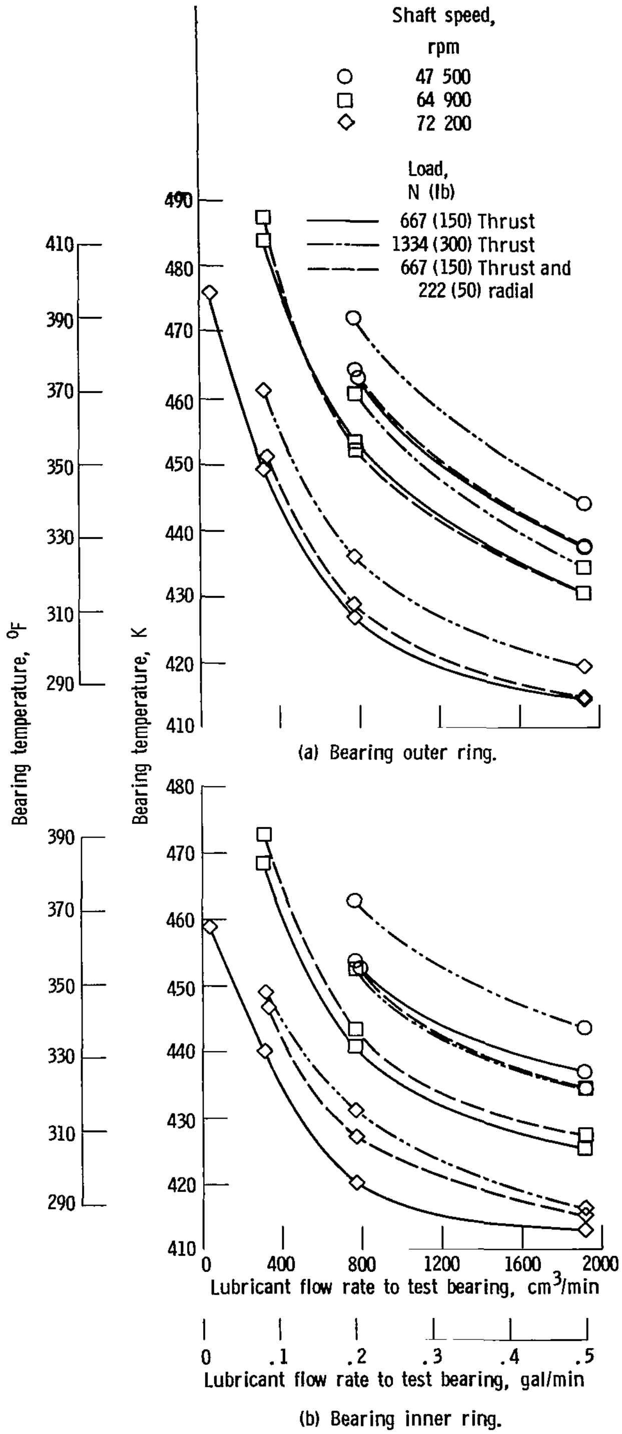


Figure 8. - Effect of lubricant flow rate on test-bearing temperature at three different load conditions. Oil-in temperature, 394 K (250° F).

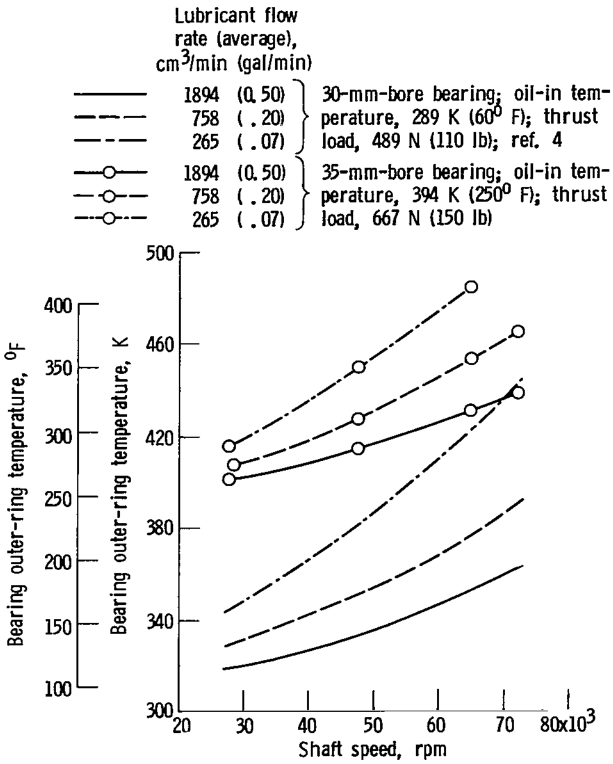


Figure 9. - Effect of shaft speed on outer-ring temperature for two different ball bearings, each with a different oil-in temperature.

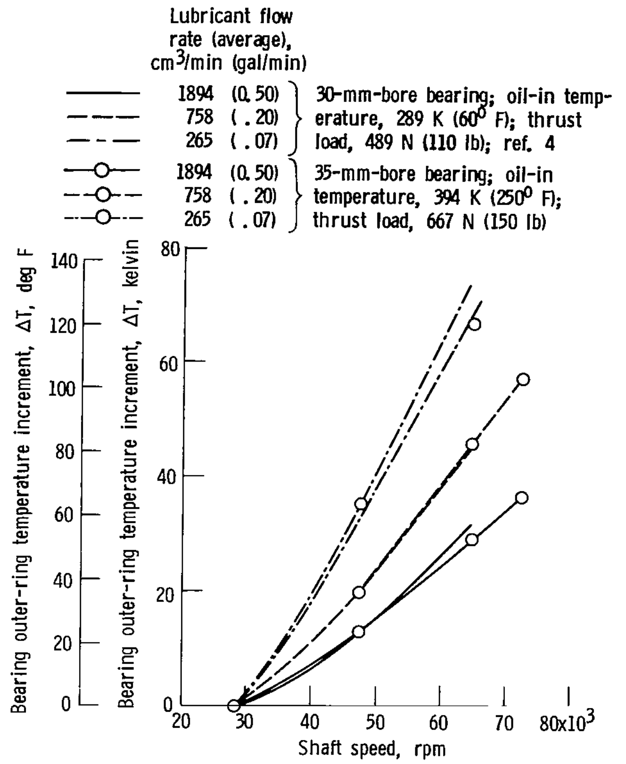


Figure 10. - Relative slopes of outer-ring-temperature-versus-shaft-speed curves for two different ball bearings, each with a different oil-in temperature.

Figure 10 shows more clearly how similar the slopes of the curves of equal lubricant flow rate are for the two bearings. These curves were plotted by using the temperature at a shaft speed of 29 000 rpm as a reference value for each curve and determining the temperature increase ΔT at various speeds from figure 9. These curves indicate that the oil-in temperature does not appreciably change the basic characteristics of the speed-versus-outer-ring-temperature curves for bearings of similar bore size.

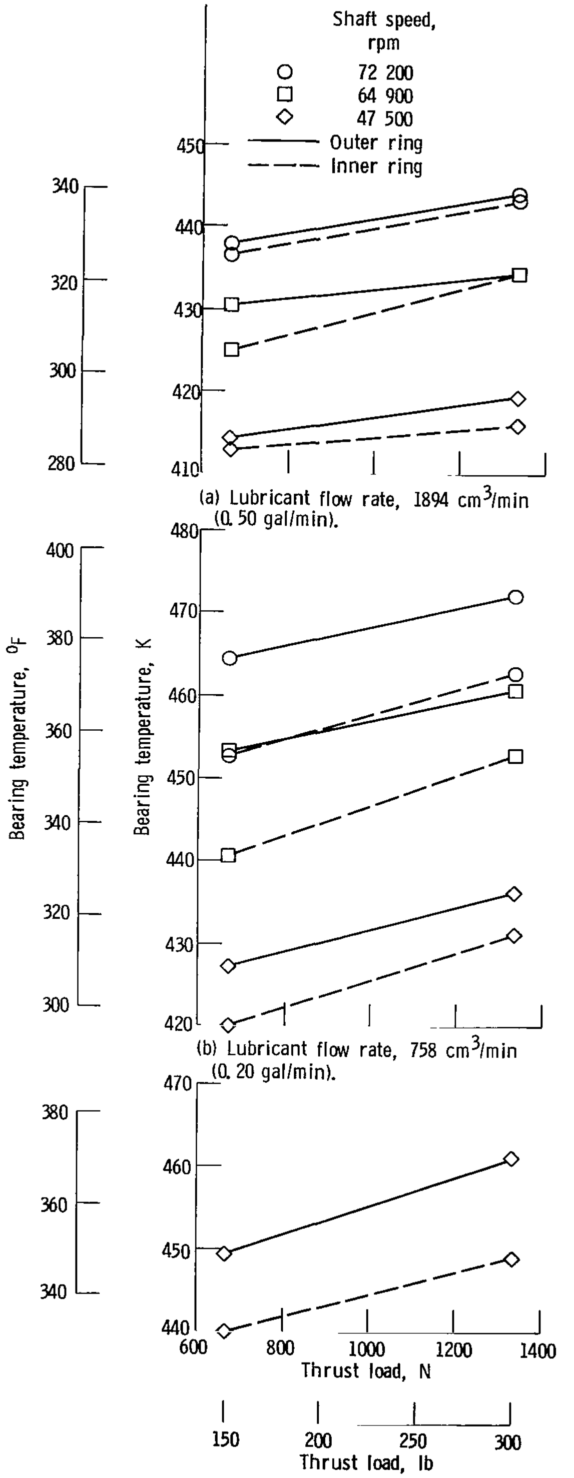
Effect of Load on Bearing Temperature

The effect of load on bearing temperature is shown in figure 11. Temperature increased with load for all values of lubricant flow rate investigated. The inner-ring temperature remained lower than the outer-ring temperature at all speeds and lubricant flow rates investigated. Doubling the thrust load, in general, had very little effect on bearing temperatures. The difference in temperature between the inner and outer rings increased with decreasing lubricant flow rate at both thrust loads tested.

Bearing Power Loss

Approaches to determining bearing power loss. – Two approaches were used to determine bearing power loss: Outer-ring bearing torque was measured, and heat rejection to the lubricant was calculated. Although the strain gage and the related torque mechanism were calibrated and checked before each test run, hysteresis losses in the knife-edges, restraints due to lubrication and thermocouple lines, and rig vibrations could cause some inaccuracies in the torque readings.

Heat rejection to the lubricant is a major portion of the bearing power loss but does not account for all the heat lost by conduction, radiation, and convection. Because of this fact and possible inaccuracies in oil-in and oil-out temperature measurements, which depend on how close the thermocouples are to the bearing, one would expect the bearing power loss calculated from measured bearing torque to be greater than that calculated from heat rejection to the lubricant. This was true, as shown in figures 12 and



(a) Lubricant flow rate, 1894 cm³/min (0.50 gal/min).

(b) Lubricant flow rate, 758 cm³/min (0.20 gal/min).

(c) Lubricant flow rate, 303 cm³/min (0.08 gal/min).

Figure 11. - Effect of load on bearing temperature. Oil-in temperature, 394 K (250° F).

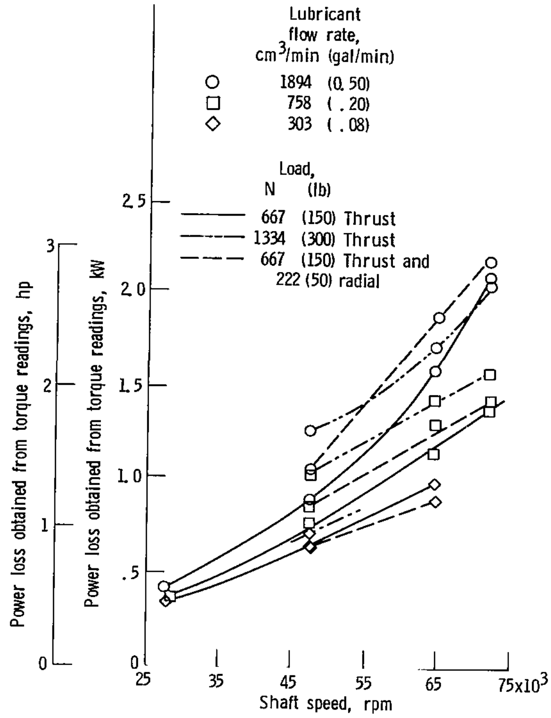


Figure 12. - Effect of shaft speed on power loss obtained from torque readings at three different load conditions. Oil-in temperature, 394 K (250° F).

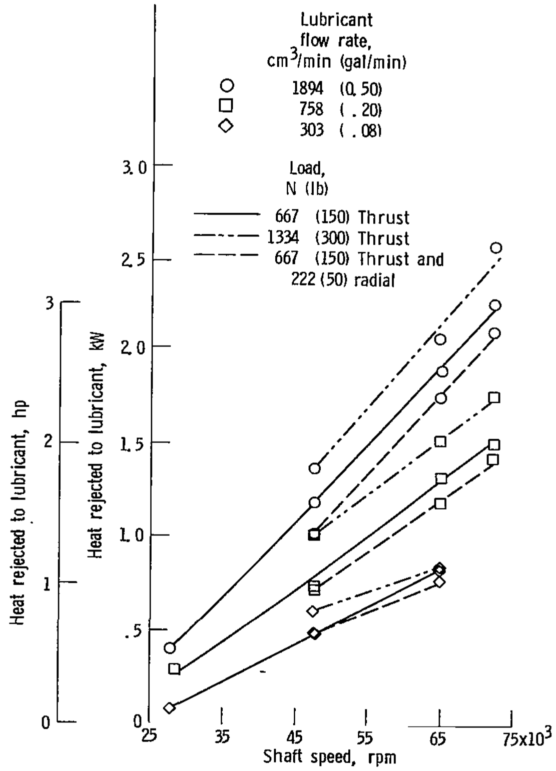


Figure 13. - Effect of shaft speed on heat rejected to lubricant at three different load conditions. Oil-in temperature, 394 K (250° F).

13, except at the highest flow rate and speed conditions. These exceptions indicate that the measured bearing torque is low because of the possible errors suggested above and/or that the heat loss other than that rejected to the lubricant is small.

So that heat rejection and thus power loss within the bearing could be measured, oil-in and -out temperatures were obtained for all flow conditions. Total heat absorbed by the lubricant was obtained from the standard heat-transfer equation

$$Q_T = MC_p(t_{out} - t_{in})$$

where

Q_T total heat-transfer rate to lubricant, J/min (Btu/min)

M mass flow rate, kg/min (lb/min)

C_p specific heat, J/kg K (Btu/lb °F)

t_{out} oil-out temperature, K (°F)

t_{in} oil-in temperature, K (°F)

Bearing power loss at three different load conditions. – The power loss in the bearing at three different load conditions determined from torque readings taken with a strain gage is shown in figure 12. Power loss increased with increasing speed and lubricant flow rate. Maximum power loss was obtained with a maximum thrust load of 1334 N (300 lb) at each flow rate, except at the maximum flow rate of 1894 cm^3/min (0.50 gal/min), beyond a speed of approximately 55 000 rpm. At 70 000 rpm and 1894 cm^3/min (0.50 gal/min) the value of power loss at 1334 N (300 lb) dropped below that at the other load conditions. Since this is a highly improbable situation, it must be assumed that the inherent inaccuracies in torque measurement mentioned previously are the cause.

The power loss values for a bearing with the combined load were higher than those for a bearing with the 667-N (150-lb) thrust load except at the lowest lubricant flow rate of 303 cm^3/min (0.08 gal/min), figure 12. Inherent inaccuracies in oil-in and oil-out temperature measurements, mentioned previously, probably resulted in the combined-load data being at a slightly lower temperature than the 667-N (150-lb) thrust load data at approximately 65 000 rpm.

The results of the heat-transfer calculations are shown in figure 13 in the form of power rejection to the lubricant versus shaft speed. (For convenience, values for heat transfer were converted from joules per minute to kilowatts.) Figure 13 shows that heat rejection to the lubricant increased as the speed, lubricant flow rate, and thrust load increased. The curves of this figure, unlike those of figure 12, are consistent in that they show that the power loss diminished at each lubricant flow rate as the load was changed from 1334 N (300 lb) thrust to 667 N (150 lb) thrust, to a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial.

For the 35-mm-bore bearing tested, the power loss in the bearing varied from 0.1 hp to 3.4 hp as the speed was increased from a nominal 28 000 rpm to 72 000 rpm at a lubricant flow range of 303 to 1894 cm^3/min (0.08 to 0.50 gal/min), at the three different load conditions used.

Cage Slip

Effect of lubricant flow rate on cage slip. – So that percent cage slip could be determined, the epicyclic cage speeds C_{epi} at the various test speed and load conditions were obtained from a computer program that took into account centrifugal force effects on contact angle. Elastic contact forces were considered in a race-control solution. Thermal and lubricant effects were not considered in this computer solution of epicyclic cage speed. The epicyclic cage speed was combined with the measured experimental cage speed C_{exp} to obtain percent cage slip as follows:

$$\text{Percent cage slip} = 1 - \frac{C_{exp}}{C_{epi}} (100)$$

The effect of lubricant flow rate on percent cage slip at three different load conditions and shaft speeds is shown in figure 14. Because negative values for percent cage slip were obtained at 47 500-rpm shaft speed and 1334-N (300-lb) thrust load, no plot of these data is shown in figure 14(a). The negative values ranged from -1.18 to -2.41 percent. These negative values are probably due to the rather simplified computer program used to obtain epicyclic cage speed, and an assumption of zero slip at these conditions might well be justified. All cage slips less than approximately 2 or 3 percent appear to be within the range of error of the calculations. The data at 1334-N (300-lb) thrust load in figures 14(c) and (b) indicate that as the speed decreased from 72 200 rpm to 64 900 rpm, the percent cage slip decreased accordingly, and therefore the assumption of zero percent slip at a further reduced speed of 47 500 rpm seems reasonable.

For all speeds tested, the increase in percent cage slip with lubricant flow rate was minimal. The addition of a 222-N (50-lb) radial load to the 667-N (150-lb) thrust load made an insignificant difference in percent cage slip. However, the percent cage slip decreased appreciably when the thrust load was doubled at shaft speeds of 64 900 and 72 200 rpm, figures 14(b) and (c), respectively. The highest percent cage slip was 6.68 and occurred at a lubricant flow rate of 1894 cm^3/min (0.50 gal/min), a shaft speed of 72 200 rpm, and a thrust load of 667 N (150 lb).

Effect of shaft speed and load on cage slip. – The effect of shaft speed on percent cage slip at each

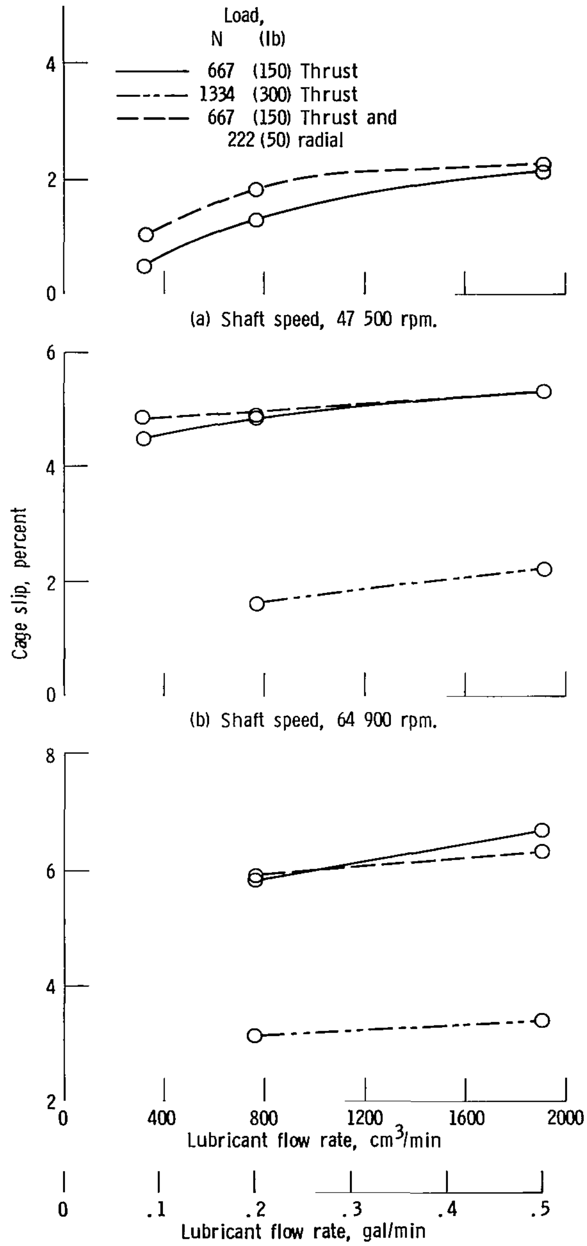


Figure 14. - Effect of lubricant flow rate on cage slip for a bearing at different load conditions. Oil-in temperature, 394 K (250° F).

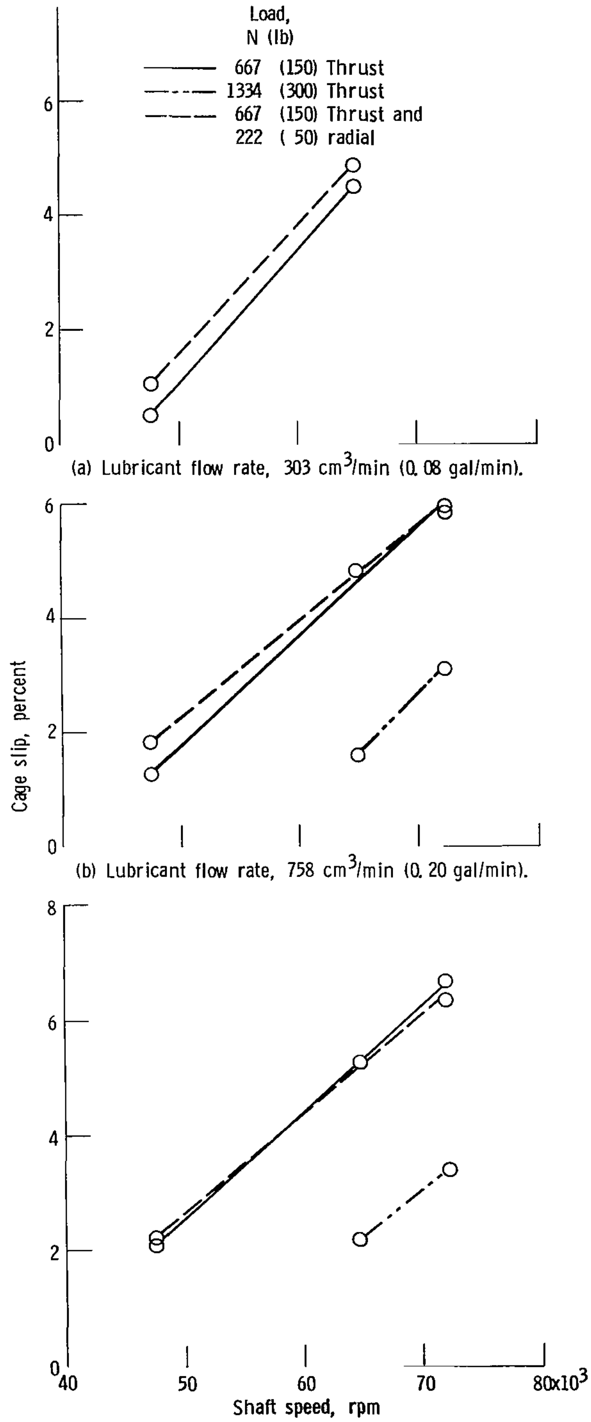


Figure 15. - Effect of shaft speed on cage slip for a bearing at three different load conditions. Oil-in temperature, 394 K (250° F).

lubricant flow rate tested and under three different load conditions is shown in figure 15. No data were obtained at a 1334-N (300-lb) thrust load and a 303-cm^3/min (0.08-gal/min) lubricant flow rate, figure 15(a), since bearing temperatures would have exceeded the arbitrarily set high limit of 491 K (425° F). Percent cage slip increased with shaft speed for each lubricant flow rate and load condition tested. Figures 15(b) and (c) again show that the addition of a 222-N (50-lb) radial load to the 667-N (150-lb) thrust load had little effect on percent cage slip, whereas doubling the thrust load decreased the percent cage slip appreciably. The greatest reduction in cage slip caused by doubling the thrust load occurred at 64 900-rpm shaft speed and 758-cm^3/min (0.20-gal/min) lubricant flow rate, where cage slip was reduced by a factor of 3.

Visual examination of the bearing after running showed no damage to the raceways or the balls. Therefore the cage slip that did occur was not of sufficient magnitude to affect the satisfactory operation of the bearing.

Concluding remarks on cage slip. – Although values of percent cage slip in these tests were generally small (<7 percent), the effects of lubricant flow rate, speed, and load were as expected. Inaccuracies in the calculation of the epicyclic cage speed probably made calculated percent cage slips of less than 2 or 3 percent meaningless. Effects of the temperature differential between the inner and outer rings, centrifugal effects of the balls, and inner-ring fits changed the contact angles at each raceway and made an accurate calculation of epicyclic cage speed very difficult. In spite of these limits the observed effect of increased lubricant flow rate was to increase the percent cage slip, as might have been expected, because of the additional drag on the balls and the cage. Likewise increased thrust load caused a decrease in percent cage slip, as was expected, from the increased traction at the ball-raceway contacts. The observed increase in percent cage slip with increased shaft speed might have been expected because centrifugal forces decrease the ball load and thus the traction at the inner-raceway contact.

Summary of Results

Parametric tests were conducted in a high-speed, high-temperature bearing tester on a 35-mm-bore, angular-contact ball bearing with a single-outer-land-guided cage. The bearing had a nominal contact angle of 24°. Provisions were made for jet lubrication of the bearing and for outer-ring cooling. Test conditions included thrust loads of 667 and 1334 N (150 and 300 lb) and a combined radial and thrust load of 222 and 667 N (50 and 150 lb), respectively. Nominal shaft speeds ranged from 28 000 to 72 000 rpm, with an oil-in temperature of 394 K (250° F). Oil was jet fed to the bearing at flow rates ranging from 76 to 1894 cm^3/min (0.02 to 0.50 gal/min) and at a jet velocity of 20 m/sec (66 ft/sec). The lubricant, neopentylpolyol (tetra) ester, met the MIL–L–23699 specifications. The following results were obtained:

1. A 35-millimeter-bore ball bearing with jet lubrication was successfully operated to 2.5 million DN at a maximum thrust load of 1334 N (300 lb) and a combined load of 667 N (150 lb) thrust and 222 N (50 lb) radial.
2. Bearing temperatures increased with shaft speed and decreased with increasing lubricant flow rate, with the inner-ring temperature generally lower than the outer-ring temperature over the speed and lubricant-flow-rate ranges investigated.
3. A bearing with a thrust load of 1334 N (300 lb) produced higher temperatures than either one with a 667-N (150-lb) thrust load or one with a combined 667-N (150-lb) thrust and 222-N (50-lb) radial load, although the effect of load was relatively small.
4. If the outer-ring temperature was greater than the inner, cooling oil flow could be employed to thermally balance the bearing. With outer-ring cooling, higher operating speeds could be achieved with lower lubricant flow rates, and this resulted in lower operating temperatures.
5. Bearing power loss increased as the lubricant flow rate, speed, or load was increased. The greatest amount of heat rejection to the lubricating oil by the bearing occurred with a 1334-N (300-lb) thrust load, followed by a 667-N (150-lb) thrust-loaded bearing and then the 667-N (150-lb) thrust, 222-N (50-lb) radial combined-loaded bearing.
6. For all speed and load conditions tested, the increase in percent cage slip with lubricant flow rate was minimal. Percent cage slip decreased significantly when the thrust load was doubled, and increased appreciably with speed for the lubricant flow rates and loads tested.
7. The oil-in temperature did not appreciably change the basic characteristics of the speed-versus-outer-ring-temperature curves when compared with data from another study for bearings of similar bore size but with room-temperature inlet oil.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 19, 1979,
505–04.

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1. Report No.
NASA TP-1657

2. Government Accession No.

3. Recipient's Catalog No.

4. Title and Subtitle
OPERATING CHARACTERISTICS OF A HIGH-SPEED, JET-LUBRICATED 35-MILLIMETER-BORE BALL BEARING WITH A SINGLE-OUTER-LAND-GUIDED CAGE

5. Report Date
April 1980

6. Performing Organization Code

7. Author(s)
Fredrick T. Schuller, Stanley I. Pinel, and Hans R. Signer

8. Performing Organization Report No.
E-220

9. Performing Organization Name and Address
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

10. Work Unit No.
505-04

11. Contract or Grant No.

12. Sponsoring Agency Name and Address
National Aeronautics and Space Administration
Washington, D.C. 20546

13. Type of Report and Period Covered
Technical Paper

14. Sponsoring Agency Code

15. Supplementary Notes
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16. Abstract

Parametric tests of a 35-mm-bore, angular-contact ball bearing with a single-outer-land-guided cage were conducted on a high-temperature, high-speed bearing tester. Provisions were made for jet lubrication of the bearing and for outer-ring cooling. Test conditions included two different thrust loads and a combined thrust and radial load. Nominal shaft speeds were 28 000 to 72 000 rpm, with an oil-in temperature of 394 K (250° F). The bearing was successfully operated to 2.5 million DN at a maximum thrust load of 1334 N (300 lb) and at a combined 667-N (150-lb) thrust and 222-N (50-lb) radial load. Bearing temperatures increased with shaft speed and decreased with increasing lubricant flow rate. Inner-ring temperature was generally lower than outer-ring temperature. Lower operating temperatures resulted from cooling the outer ring. Bearing power loss increased as the lubricant flow rate, speed, or load was increased. The increase in percent cage slip with increasing lubricant flow rate was minimal for all speed and load conditions tested. Percent cage slip decreased significantly when the thrust load was doubled, but it increased appreciably with speed.

17. Key Words (Suggested by Author(s))
High-speed bearings; Rolling-element bearings; Bearings; Ball bearings; Bearing lubrication; Jet-lubricated high-DN bearings

18. Distribution Statement
Unclassified - unlimited
STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	15	A02

* For sale by the National Technical Information Service, Springfield, Virginia 22161

NASA-Langley, 1980

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